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AN ANALYTICAL INVESTIGATION OF A CONCEPTUAL DESIGN
FOR THE SPACE STATION TRANSVERSE BOOM ROTARY JOINT
STRUCTURE

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National Aeronautics and
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AN ANALYTICAL INVESTIGATION OF A CONCEPTUAL DESIGN FOR THE
SPACE STATION TRANSVERSE BOOM ROTARY JOINT STRUCTURE

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SUMMARY

A study was conducted to define an annular ring, discrete roller assembly concept for the design of the Space Station transverse boom rotary joint. The concept was analyzed using closed-form and finite element techniques, to size structural members for a range of joint diameters and to determine necessary equivalent stiffnesses for roller assemblies. Also, a mass study of the system was conducted to determine its practicality, and maximum member loads in the joint were identified. All analyses were based on the reference Space Station Initial Operating Capability (IOC) configuration.

INTRODUCTION

NASA has defined a reference configuration for Space Station (see figure 1) which is a gravity gradient stabilized design using either photovoltaic or solar dynamic methods for solar energy conversion (ref. 1). To allow for a continuous correction for the solar incidence angle, these power conversion devices are to be structurally supported with rotational degrees of freedom in two orthogonal directions.

Figure 2 shows the orbital orientation of Space Station in the gravity gradient stabilized attitude, with its central keel beam aligned with the nadir. Also depicted in this figure are the two solar array rotational degrees of freedom, the alpha and beta angles. The alpha rotation is accommodated by two identical large rotary joints in the truss structure of the transverse boom.

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This rotary joint must provide stiffness and strength equivalent to the parent truss structure of the transverse boom. In addition, the joint must rotate more than 180° on the sunny side of the orbit, and either rewind or rotate continuously on the shady side of the orbit to reposition the solar wings for the next orbit. Finally, the joint must permit on-orbit assembly, growth, and maintenance over a long lifetime.

Four primary truss concepts for space station were evaluated in reference 2. The results of this trade study indicate that a 15 foot square bay truss is preferred on the bases of growth potential and customer accommodations. This truss is the largest and the stiffest of the concepts considered, and there is concern that a rotary joint designed to maintain this truss stiffness would be prohibitively heavy. This paper presents a concept for such a joint, parametric analyses of the concept, and a mass study of the rotary joint structure.

SYMBOLS

α	=	alpha rotation angle of solar wing
E	=	Young's modulus
G	=	shear modulus
I	=	moment of inertia
J	=	torsion constant
A	=	area
P	=	annular ring load
Δp	=	deflection in annular ring due to load P
R	=	radius of annular ring
M_x	=	resultant internal bending moment in annular ring
M_z	=	resultant internal torsion moment in annular ring
P_ϕ	=	resultant internal shear force in annular ring

ϕ	= arc angle
P_0	= boundary force on arc segment of ring
M_{x0}, M_{z0}	= boundary moments on arc segment of ring
t	= thickness of cross section of ring
U	= internal strain energy

CONCEPT DEFINITION

The rotary joint concept to be studied is shown in figure 3. The rotational interface consists of an annular ring and roller track connected to the 15 foot box truss by a system of 24 transition truss members, 12 inboard and 12 outboard of the ring. The transition members terminate at eight discrete attachment points on either side of the ring, forming regular octagons around the ring. The inboard transition members are solidly fixed to the ring. The outboard transition members are fixed to eight discrete roller assemblies which are, in turn, mounted to the roller track portion of the annular ring. These assemblies are also interconnected with cross tie members which maintain spacing and transmit circumferential loads between assemblies. Each roller assembly transmits radial loads and thrust loads to the ring (thrust loads are normal to the plane of the ring). Also, one of the roller assemblies incorporates a drive mechanism to rotate the joint and transmit circumferential loads to the ring. The rotary joint occupies two bays of the 15 foot truss beam.

Figure 4 shows a structural characteristic of this conceptual joint design that arises from the discrete bearing approach. For $\alpha = 0^\circ$ and multiples of 45° , transition truss attachment points are aligned across the ring. However, for all rotation angles in between, the transition struts are not aligned across the ring and load transfer across the ring will involve bending and torsion of the ring. Subsequent analyses will examine the structural performance of the rotary

joint concept presented for the two cases, $\alpha = 0^\circ$ and $\alpha = 22.5^\circ$. The study will examine the influence that the joint rotation angle and the various system design parameters have on structural performance.

ANALYSIS OF THE ANNULAR RING

Internal forces in the transition truss members will be generated during bending and torsional vibration of the transverse boom. As the boom rotates the loading condition on the annular ring will change. For a highly flexible ring, the stiffness of the rotary joint system will vary greatly with rotation angle and there will be a corresponding variation in the vibration frequencies of the transverse boom. Therefore, the ring should be designed to eliminate, within practical limits, the variation of structural performance with rotational position.

To study the elastic response of the ring, a static ring deflection analysis was conducted for the external loading condition shown in figure 5. The eight inboard attachments of the annular ring were considered to be simple supports and the eight roller assembly reactions were considered to be equal forces (P), all perpendicular to the plane of the ring. This is an approximation of the load condition that the ring would see for $\alpha = 22.5^\circ$ and the transverse boom being compressed by a force of $8(P)$. Due to symmetry, it is only necessary to consider $1/8$ of the annular ring, the 45° arc section connecting any two adjacent inboard simple supports.

For this study, an energy method approach for the static analysis of curved beams was used (ref. 4). The deflection of a linearly elastic curved beam of radius R and arbitrary cross section (EI , EA , GJ) at the point of application of a load P was found by deriving an expression for the total internal strain energy of the curved beam and applying Castigliano's theorem.

Neglecting transverse shearing, the expression for the total internal strain energy of an arc segment of the ring (seen in figure 6) is:

$$U = \int_{\phi} \left\{ \frac{(M_x)^2}{2EI} + \frac{(M_z)^2}{2GJ} \right\} R d\phi \quad (1)$$

where M_x and M_z are resultant internal bending and torsion moments, respectively. It will be shown that these quantities are functions of the applied load P and the angle ϕ only. Also shown in figure 6 are boundary loads, P_0 , M_{x0} , and M_{z0} applied to the arc segment to represent the removed portion of the ring, and P_ϕ , the resultant internal shear force in the ring. For simplicity, the ring is assumed to be loaded through its elastic axis. This assumption is justified in Appendix A.

Equilibrium of forces for the arc shows that $P_\phi = P_0 = P/2$, for $\alpha = 27.5^\circ$. Equilibrium of moments at any arbitrary location along the arc, given by ϕ , results in expressions for M_x and M_y .

For the range, $0 \leq \phi \leq 22.5^\circ$:

$$\begin{aligned} M_x &= M_{x0} \cos \phi - M_{z0} \sin \phi - PR/2 \sin \phi \\ M_z &= M_{z0} \cos \phi + M_{x0} \sin \phi - PR/2 (1 - \cos \phi). \end{aligned} \quad (2)$$

For the range, $22.5 \leq \phi \leq 45.0^\circ$:

$$\begin{aligned} M_x &= M_{x0} \cos \phi - M_{z0} \sin \phi - PR/2 \sin \phi + PR \sin (\phi - \pi/8) \\ M_z &= M_{z0} \cos \phi + M_{x0} \sin \phi - PR/2 (1 - \cos \phi) + PR (1 - \cos(\phi - \pi/8)). \end{aligned} \quad (3)$$

Using these expressions and satisfying the symmetry condition that the boundary loads at $\phi = 0$ equal the boundary loads at $\phi = 45^\circ$ results in:

$$M_{z0} = 0, \quad M_{x0} = .09945(PR)$$

The equations for M_x and M_z can be expressed in terms of the variables P and ϕ , and the internal strain energy can be found by substituting these quantities into equation (1) and integrating over the limits of ϕ from 0° to 45° . This will give:

$$U = \frac{P^2 R^3}{EI} (1.30 \times 10^{-3}) + \frac{P^2 R^3}{GJ} (2.03 \times 10^{-5}) \quad (4)$$

Applying Castigliano's Theroem (ref. 4) yields deflection of the curved beam at the point of application of the bearing load P , in the direction of the load. or:

$$\Delta_p = \frac{\delta U}{\delta P}$$

which results in:

$$\Delta_p = PR^3 \left\{ \frac{2.6 \times 10^{-3}}{EI} + \frac{4.1 \times 10^{-5}}{GJ} \right\}. \quad (5)$$

The deflection of the annular ring increases with the cube of its radius, and depends both on its bending stiffness and torsional stiffness. However, the contribution due to torsion is about two orders of magnitude less than that for bending, provided the torsional stiffness is equal to or greater than the bending stiffness. This result has implications for design of the ring cross section. The bending moment of inertia (I) can be studied alone if the torsional constant (J) is large enough to neglect torsional effects.

Figure 7 shows the general ring cross section that is considered herein. It is a square box, six inches by six inches with a variable thickness t . The expressions for the moment of inertia and the torsional constant as functions of the thickness, t , are:

$$\begin{aligned}
 I &= \frac{(6+t)^4 - (6-t)^4}{12} \\
 J &= (6-t)^3 t
 \end{aligned}
 \tag{6}$$

It can be shown that the torsional constant is greater than the moment of inertia for values of t less than about 0.7 inches. The cross sectional geometry presented in figure 7 is not necessarily intended to be indicative of an actual ring design. This geometry is considered to give physical significance to the values of the cross sectional parameters (I , J) and provide an estimate of ring size for ring mass calculations. For the analysis herein, it is assumed that the ring has this cross section and is made of aluminum ($E = 10 \times 10^6$ psi, $\rho = .1$ lb/in³).

FINITE ELEMENT MODELS

A linear finite element analysis of the rotary joint configuration was conducted. Figure 8 shows three models of the space station transverse boom that were used for this analysis. These include one model of the boom cantilevered without a rotary joint, and two models of the boom cantilevered with the rotary joint included, one for the joint at $\alpha = 0^\circ$ and the other for the joint at $\alpha = 22.5^\circ$. The model of the boom without the joint is included to provide a basis of comparison of the stiffness of the rotary joint assembly and the parent truss.

In these models, the truss struts were axial elements representing graphite/epoxy tubes, two inches in diameter, with a cross sectional area of .3657 in², a Young's modulus of 40×10^6 psi ($EA = 1.46 \cdot 10^7$ lb), and a density of .058 lb/in³. Rigid masses were attached at the nodes to represent joints, and the joints were assumed to have the same stiffness as the struts. The solar arrays were modeled as rigid beams with outrigger supports having masses scaled from

the Solar Array Flight Experiment (SAFE) as indicated in reference 5. The arrays were assumed to be rigid to eliminate solar blanket vibrations from the analysis. The transition truss members of the rotary joint were assumed to be graphite/epoxy elements also but with arbitrary cross sectional area. The annular ring was modeled as a 6 x 6 inch aluminum box beam with a thickness which will be varied. Finally, the outboard transition truss was attached to the ring with three mutually perpendicular linear springs having variable stiffnesses to represent the radial, drive and thrust stiffness components of the roller assemblies.

A number of non-structural components were not considered in these models. Among them are, masses of the roller assemblies and masses and locations of power coupling, fluid coupling, drive system hardware, and utility lines.

FINITE ELEMENT ANALYSIS

The primary design goal for the rotary joint is to maintain the stiffness of the parent 15 foot truss. A number of variables exist that affect this stiffness: (1) ring cross section, (2) ring diameter, (3) transition truss member stiffness, and (4) roller assembly stiffness. A reduction of the rotary joint stiffness will reduce the vibration frequencies of the transverse boom. Therefore, studying the effect these parameters have on reducing boom frequencies will identify their effect on overall rotary joint stiffness. The first bending and first torsional frequency of the wing were calculated for ranges of each of the variables. This information was used to arrive at acceptable values for these variables.

Sizing of Annular Ring Thickness

The primary cause of variations in the transverse boom frequencies as a function of the alpha rotation angle would be flexibility of the annular ring. As indicated in equation (5), for a fixed ring cross section, the deflection is greatest for the largest ring diameter. By studying the maximum diameter system and choosing the ring cross sectional size that will result in the transverse boom frequencies being essentially invariant with rotation angle, alpha, it follows that this thickness would be adequate for smaller diameter rings. Because the usable cargo envelope for the shuttle is slightly less than 15 feet in diameter, a maximum diameter of 14 feet was considered for the rotary joint annular ring, as it was assumed that the annular ring, roller assemblies, drive unit, and a power transfer device would be carried to orbit pre-assembled.

Assuming 14 feet for the diameter of the annular ring, $.3657 \text{ in}^2$ for the cross sectional area of the transition members, (same as regular truss members), and $10^9 \text{ lb}_f/\text{in}$ (essentially infinite rigidity) for the roller assembly stiffness components, the first bending and first torsional frequencies of the transverse boom were calculated using the finite element models defined, for $\alpha = 0^\circ$ and 22.5° . These results are shown in figure 9 as a function of the thickness (t) of the annular ring cross section. The first bending ($f = .9252 \text{ hz}$) and first torsion ($f = .6919 \text{ hz}$) frequencies of the transverse boom with no rotary joint are also shown in figure 9. As the thickness and, correspondingly, the bending stiffness of the ring is increased, the frequencies at $\alpha = 22.5^\circ$ (maximum elastic deformation in ring) approach the frequencies at $\alpha = 0.0^\circ$ (minimum elastic deformation in ring). At a cross sectional thickness of $1/4$ inch, the first bending frequency of the wing at alpha equal to 22.5 degrees is less than five percent below the first bending frequency at alpha equal to 0.0 degrees, and the first torsional frequencies are virtually equal. For thicknesses

larger than 1/4 inch, the bending frequency increases very slowly, whereas for thicknesses much less than 1/4 inch, both the bending and torsion frequencies begin to drop off fairly rapidly. For this study, 1/4 inch was chosen for the thickness of the annular ring.

Effect of Transition Truss Member Stiffness and Ring Diameter

With essentially a rigid annular ring and roller assemblies, the stiffness of the rotary joint is attributable to the geometry of the transition truss, which changes with ring diameter, and the extensional stiffness (EA) of its members. As the diameter is reduced, the depth of the truss decreases as do the angles the truss members make with the plane of the ring. To regain the truss stiffness lost by a reduction in the ring diameter, it is necessary to increase the extensional stiffness, or cross sectional area of the transition truss members.

Figure 10 shows the effect of ring diameter and transition truss member stiffness on the first bending and torsional frequencies of the transverse boom for $\alpha = 0^\circ$. In this figure, the actual frequencies are normalized to design frequencies and plotted for ranges of the two variables. The design frequencies that were used are $f_1 = .6919$ hz and $f_2 = .9252$ hz. These frequencies correspond to the first torsion and first bending modes of the transverse boom where the rotary joint has been replaced with the parent 15 foot truss (see figures 8 and 9).

For a stiff rotary joint, f_1 is the first torsional mode of the wing and f_2 is the first bending mode of the wing. However, as the ring diameter and/or the truss member stiffness are reduced, both frequencies decrease and the mode shapes couple, and then switch. For the more flexible rotary joint, f_1 is the first bending and f_2 is first torsion, and for intermediate stiffness both f_1 and f_2 are coupled bending-torsion. This effect is shown in figure 10.

An acceptable design for the rotary joint would display a combination of ring diameter and transition truss stiffness that results in the frequencies of the boom being equal to the design values (ie, the normalized frequencies equal one). These design points are determined from figure 10 as the intersection of the curved lines with the horizontal lines corresponding to $f/f_{\text{design}} = 1.0$. For example, referring to figure 10a, an acceptable joint design based on f_1 with a ring diameter of 12 feet would need transition truss members with an extensional stiffness of slightly less than $1.46 \cdot 10^7$ lb.

Comparing design points based on f_1 (figure 10b) to the design points based on f_2 (figure 10a), it is shown to be impossible to have a combination of ring diameter and truss member stiffness that results in both frequencies being equal to their design values simultaneously. This is due to the fact that given a rotary joint design that matches the bending stiffness of the parent truss (i.e., bending frequency), the torsional stiffness of the joint is greater than that of the parent truss (i.e., torsional frequency). A conservative design approach would match the bending stiffness of the parent truss (figure 10b) and the resulting design will, therefore, be more stiff in torsion than necessary. These design values are given in figure 11 where transition truss member stiffness is plotted against ring diameter. It is noted that, as the ring diameter decreases, the necessary stiffness of the transition truss members increases rapidly and at small ring diameters the transition truss member stiffness needed would be prohibitively large. Therefore, for this study, diameters below eight feet will not be considered.

Rotary Joint Structural Mass

Using the transition truss cross sectional area given in figure 11 (assuming $E = 40 \times 10^6$ lb/in²), and ring design parameters given in figure 9, the total structural mass of the rotary joint may be calculated as a function

of ring diameter. Figure 12 shows the variation of mass with ring diameter of the transition truss, the annular ring, and the total system. Due to the fixed cross section of the aluminum ($\rho = 0.1 \text{ lb/in}^3$) ring, its mass varies linearly with diameter. The mass of the graphite/epoxy ($\rho = 0.058 \text{ lb/in}^3$) transition truss is dependent on both the length of the truss members, as determined by joint diameter, and the cross sectional area of the truss members (see figure 11). The total mass shown is the sum of these two values, and does not include any non-structural masses such as roller assemblies, thermal control, or power transfer hardware which will probably be approximately constant with diameter. For comparison purposes, the mass of two bays of the parent 15 foot truss that was removed to accommodate the rotary joint assembly is shown.

Figure 12 shows that structural mass decreases with increasing joint diameter up to approximately 14 feet. This trend is dominated by the mass of the transition truss and, as previously discussed, decreasing the diameter very much below 8 feet will cause the transition truss mass and, correspondingly, the total structural mass to become excessively large.

It is possible to reduce the rotary joint structural mass by relaxing the joint stiffness requirement. Figure 13 shows the effect that reducing the frequency ratio (ratio of the first bending frequency to the design frequency - .9252 hz) has on rotary joint structural mass. With the ring size fixed, a reduced stiffness requirement will lower the mass of the transition truss. To design a joint with a diameter less than 14 feet, but with no more structural mass than the 14 foot joint (approximately 550 lbm), it would be necessary to compromise the stiffness of the joint and reduce the vibration frequencies of the transverse boom. For example, a 7 foot diameter rotary joint with a mass of 550 lbm would drop the first bending frequency of the transverse boom approximately 20% below the design value ($f/f_{\text{design}} = 0.8$). Therefore, no apparent

useful purpose is served by considering smaller diameter, reduced performance rotary joints, when the large diameter ($\sim 14'$), full performance rotary joint is also the minimum mass design.

Effects of Roller Assembly Stiffnesses

Another design feature that will affect the stiffness of the rotary joint is the stiffness of the roller assemblies that tie the outboard transition truss to the annular ring. These assemblies are envisioned to be rollers supported in housings that tie the outboard transition truss to the roller track. These housings will transmit loads from the truss members to the rollers and allow the rollers to be pre-loaded onto the roller track. Each of the components of the assemblies will contribute to an overall stiffness for the assembly and may exhibit some nonlinear characteristics. Without a detailed design of the roller assembly, the stiffness contributions due to individual components cannot be quantified. Therefore, it is assumed that the assemblies are linearly elastic with three mutually orthogonal stiffnesses: (1) normal to the ring plane, (2) tangent to the ring, and (3) parallel to the radius of the ring. These stiffness components are referred to as: thrust, drive, and radial, respectively (see figure 14), and to this point they have been assumed to be equal to 10^9 lb_f/in. To identify any effects the roller assembly (stiffness) may have on the vibration frequencies of the transverse boom, the magnitude of each component is varied independently while the other components remain equal to 10^9 lb_f/in.

Roller Assembly Thrust Stiffness

Figure 15 shows the effect that roller assembly thrust stiffness has on the first bending and first torsional frequency of the transverse boom for two values of rotary joint diameter (14 feet and 8 feet). These curves were generated assuming the other two roller assembly stiffness components were

equal to 10^9 lb_f/in. Decreasing the thrust stiffness of the rollers reduces the bending stiffness of the joint, and consequently, the bending frequency of the wing. To maintain the first bending frequency, the thrust stiffness would have to be approximately 0.5×10^6 lb_f/in for the 14 foot diameter joint, and approximately 2×10^6 lb_f/in for the 8 foot diameter joint.

Reducing the bending stiffness of the rotary joint also affects the torsion frequency of the transverse boom. The orthogonal tetrahedral truss configuration used as the parent truss exhibits a coupling between bending and torsional deformations due to location of the diagonal members (ref. 6). Therefore, torsional vibrations in the wing are accompanied by slight bending deformations, and reducing the bending stiffness of the rotary joint effectively decouples the torsional deformations from the bending deformations, and consequently, raises the torsional frequency.

Roller Assembly Drive Stiffness

The drive stiffness component is attributed to the rotary joint drive mechanism (anticipated to be a traction or gear drive device) which, for this study, is assumed to be located at one of the roller assemblies (see figure 14). Figure 16 shows the effect that the stiffness of this drive unit has on the boom frequencies. As expected, reducing the drive stiffness reduces the torsional stiffness of the rotary joint and the torsional frequency of the boom. The values for the drive stiffness necessary to maintain the boom torsional frequency are shown to be close to the values needed for the bearing thrust stiffness (0.5×10^6 lb_f/in for the 15 foot ring, 2×10^6 lb_f/in for the 8 foot ring). It is conceivable, however, that multiple drive units (which are desirable) could work in unison from more than one roller assembly, and thus increase the torsional stiffness of the rotary joint without increasing the stiffness of each individual drive unit.

Roller Assembly Radial Stiffness

The final roller assembly stiffness component to be considered is the radial stiffness. Figure 17 shows the effect that this stiffness component has on the boom frequencies. Since this configuration has only one drive location, torsional forces in the rotary joint are resolved into a drive force at the drive roller assembly, and radial forces at the other roller assemblies. Therefore, reducing the radial stiffness of the roller assemblies reduces the torsional stiffness of the rotary joint and the torsional frequency of the transverse boom. The radial stiffnesses that are needed to maintain the torsional frequency for this rotary joint/drive configuration are roughly 0.1×10^6 lb_f/in for the 14 foot ring, and 0.5×10^6 lb_f/in for the 8 foot ring. Providing a second drive unit opposite to the first would allow torsional forces to be carried by the drive mechanisms only and would minimize the effect of radial stiffness in the roller assemblies.

Maximum Internal Forces in Rotary Joint

The primary loads that occur at any point within the space station structure are due to the structure's response to transient external forces. In reference 1, a transient response analysis was conducted on a model of the 9 foot reference space station subjected to a series of possible external forces. Maximum internal forces were calculated for various critical locations throughout the station's structure including the transverse boom rotary joint. From this analysis it was predicted that the maximum bending moment that would occur at the rotary joint would be approximately 3,200 ft·lb, while the maximum torsional moment would be approximately 2,500 ft·lb. These loads were predicted for a shuttle docking maneuver.

These forces were statically applied to the present rotary joint concept for the range of ring diameters to calculate the resulting maximum internal

loads. Plotted in figure 18 are the maximum compressive loads predicted for any single transition truss member, as well as the maximum roller assembly thrust load, (i.e. the maximum ring load occurring normal to its plane of curvature). The transition truss strut compression loads were found to be at least an order of magnitude less than the Euler buckling load of the member for all designs considered over the range of joint diameters. The maximum roller assembly thrust loads were found to be less than 200 pounds for the 14 foot diameter ring, and increase, linearly at first, as the joint diameter is decreased.

CONCLUDING REMARKS

An annular ring, discrete roller assembly concept for the Space Station transverse boom rotary joint was studied to determine whether this concept could be designed to maintain the stiffness of the parent truss with reasonable structural mass. Design parameters were identified and analyses conducted to investigate their effects on rotary joint stiffness. Values for these parameters were selected to provide acceptable rotary joint designs. The corresponding mass of these designs was determined for a range of ring diameters. Additionally, the effects of roller assembly stiffnesses were studied and the maximum predicted internal forces in the rotary joint were quantified.

To obtain the optimum balance between high stiffness and low structural mass in the design of the rotary joint, it is necessary to maximize the diameter of the annular ring within operational constraints (i.e. shuttle cargo bay size). Further, a rotary joint designed with the largest possible ring diameter will result in minimum operational loads in both the roller assemblies and the transition truss members while also allowing minimum design stiffnesses for the roller assemblies.

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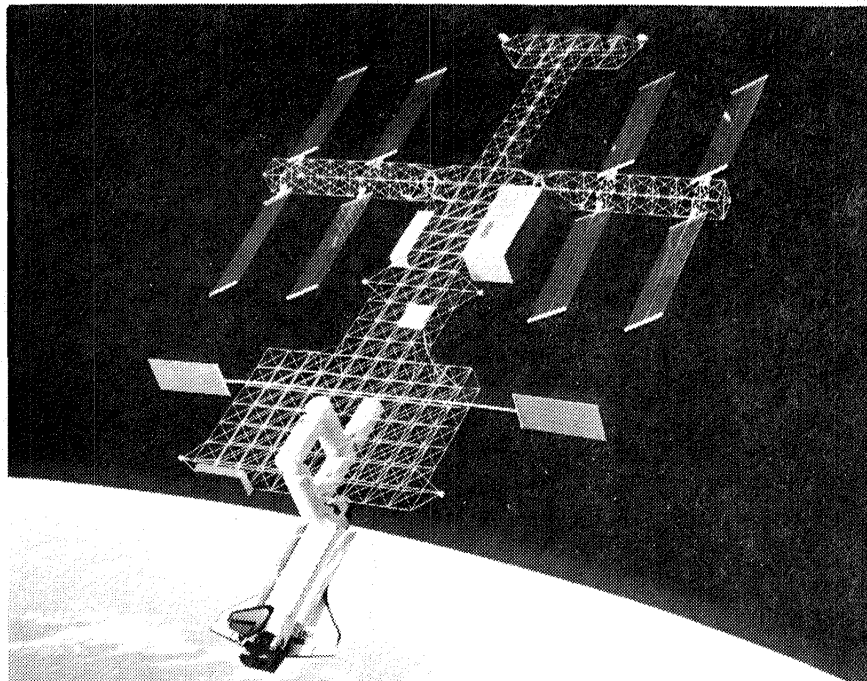


Figure 1. Space Station Reference Configuration

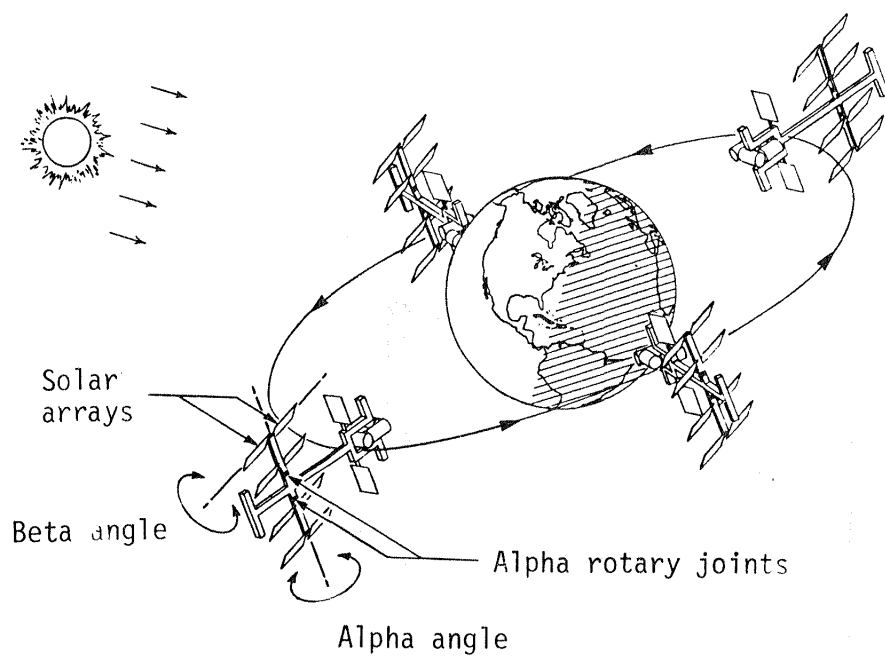


Figure 2. Orbital Orientation

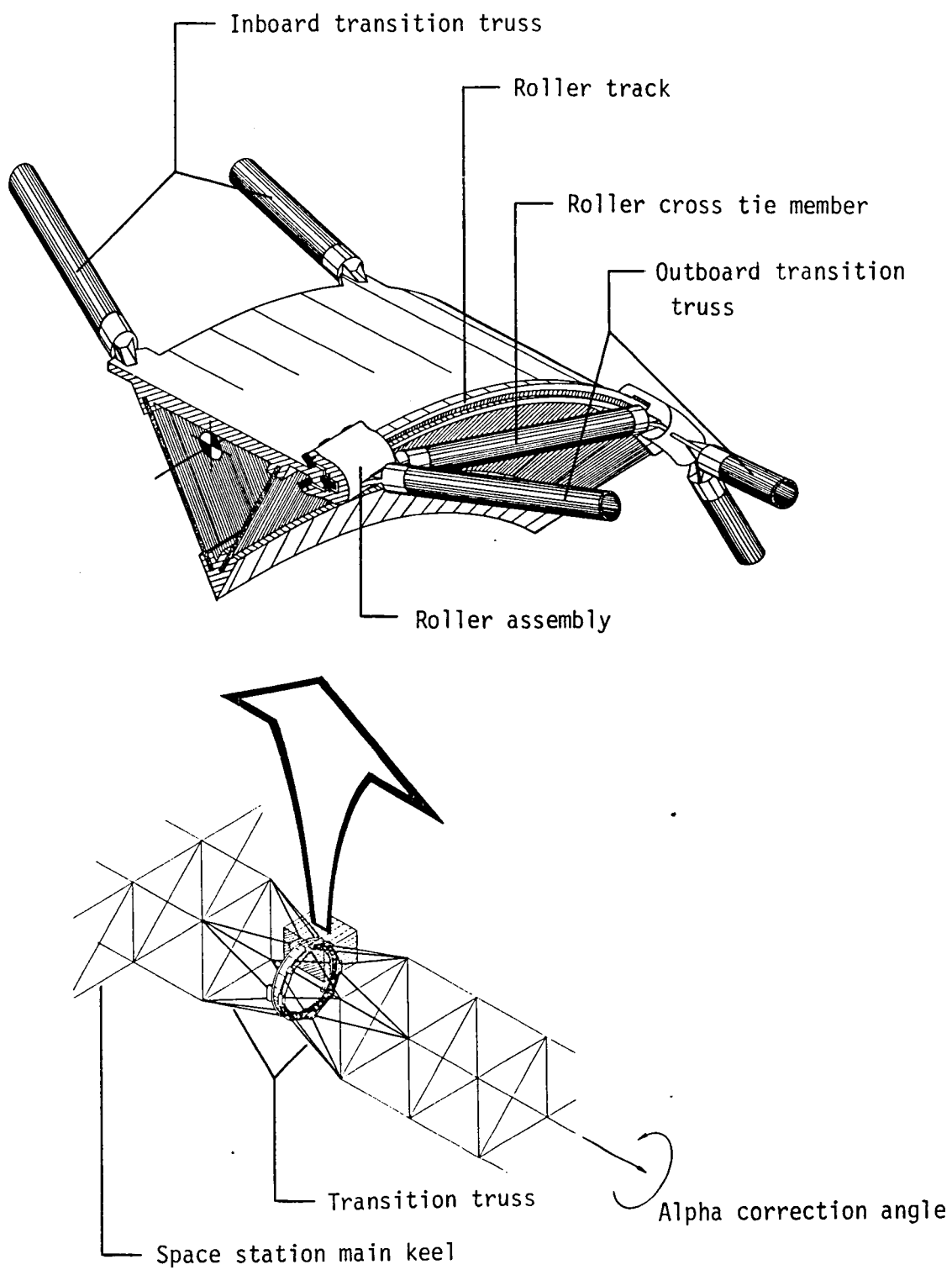


Figure 3. Alpha Rotary Joint Concept

SIDE VIEWS OF ROTARY JOINT

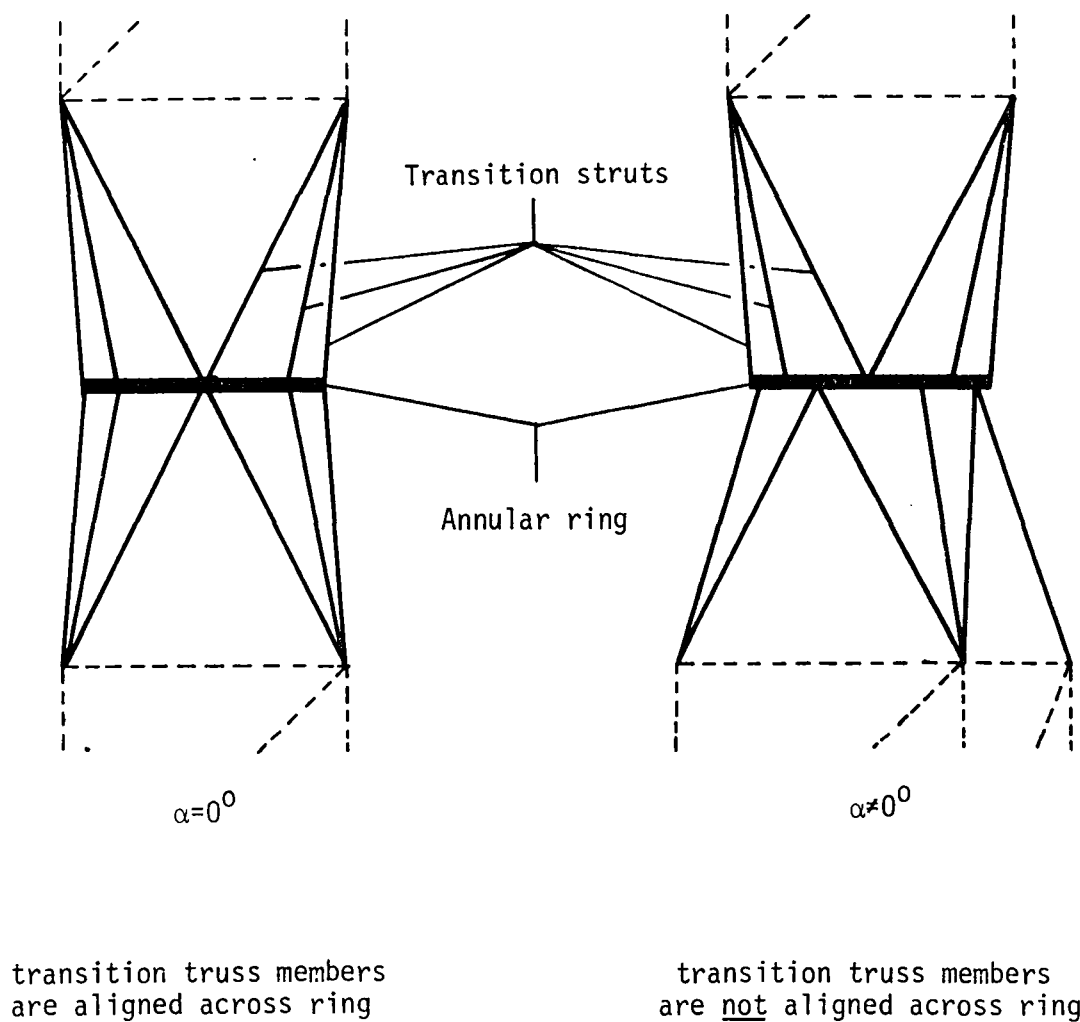


Figure 4. Annular Ring Loading Due To Discrete Bearing Approach

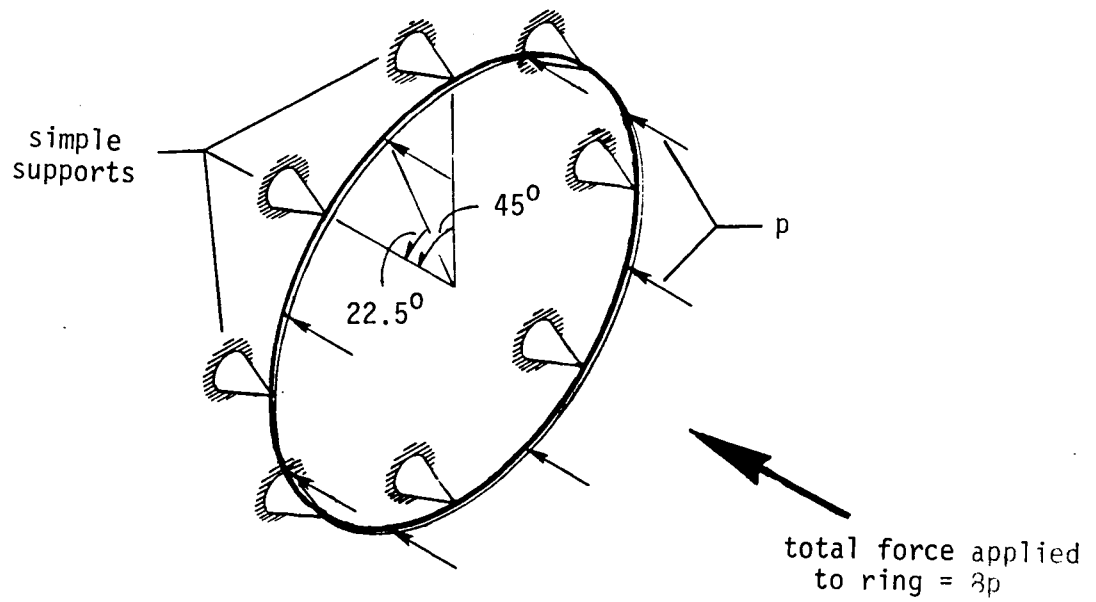


Figure 5. Simplified Ring Load Condition

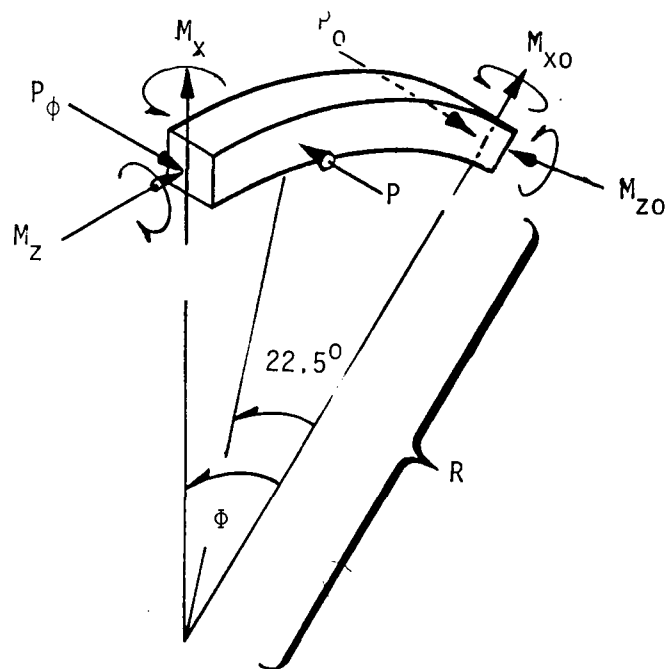
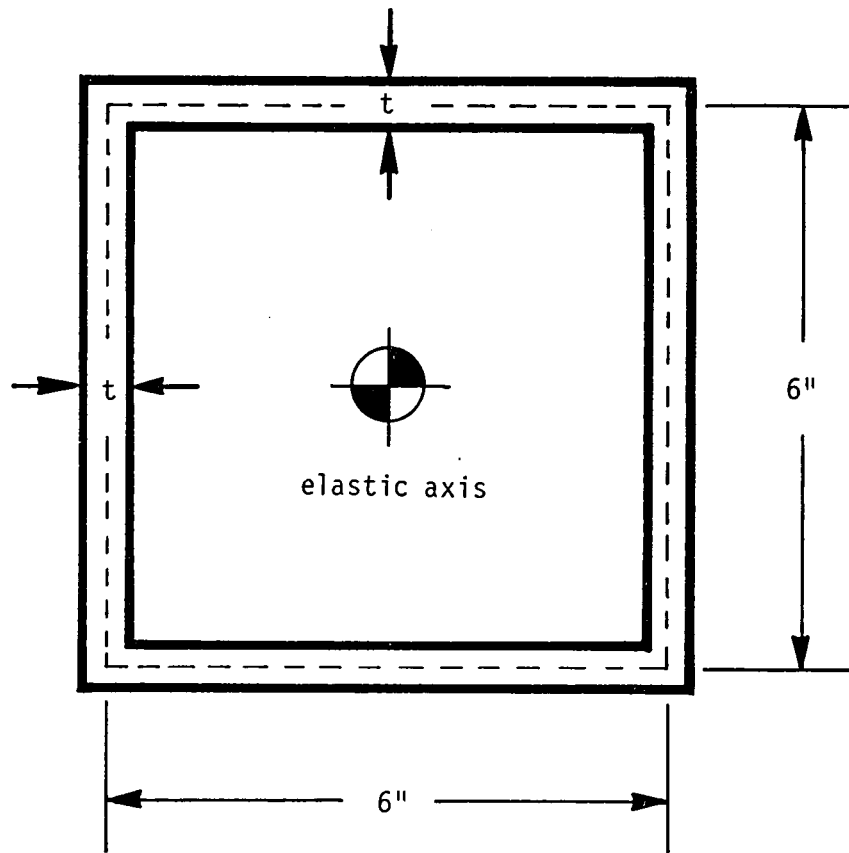


Figure 6. Internal Moments And Forces On Ring

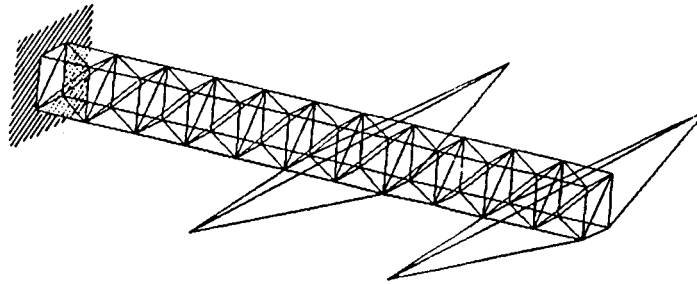


$$\text{Area} = 4(6t) \text{ in}^2$$

$$\text{Moment of Inertia} = \frac{(6+t)^4 - (6-t)^4}{12} \text{ in}^4$$

$$\text{Torsion Constant} = (6-t)^3 t \text{ in}^4$$

Figure 7. General Cross Section For Annular Ring



NO ROTARY JOINT

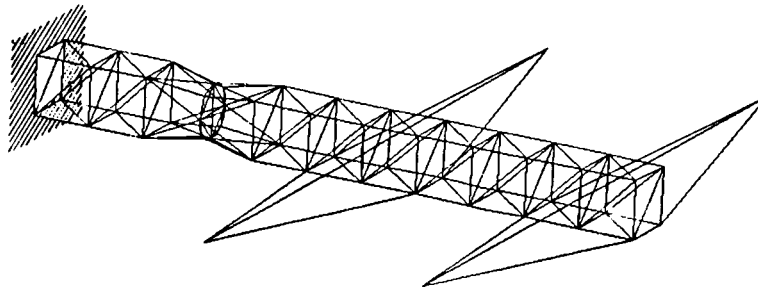
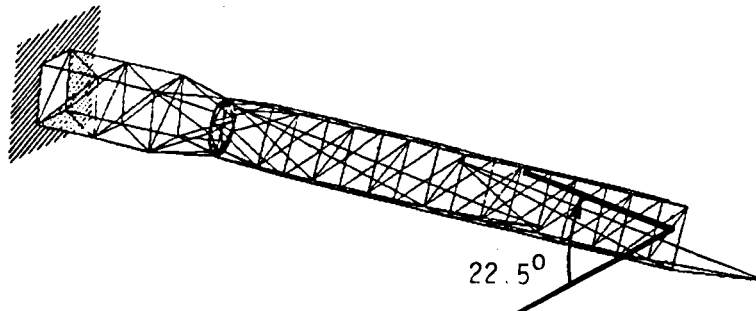
ROTARY JOINT AT $\alpha=0^\circ$ ROTARY JOINT AT $\gamma=22.5^\circ$

Figure 8. Finite Element Models Of Transverse Boom

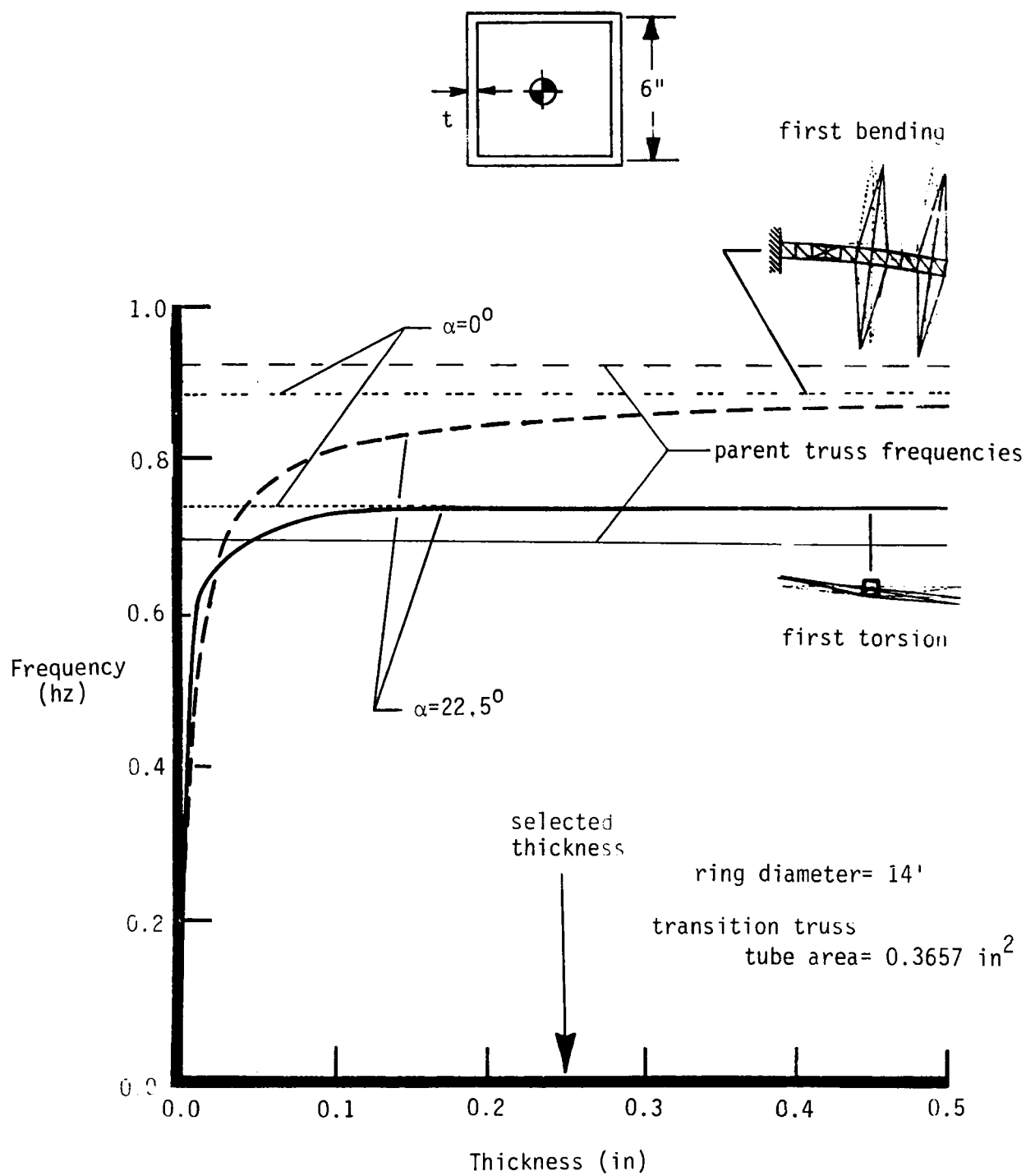


Figure 9. Effect Of Ring Thickness On Boom Frequencies

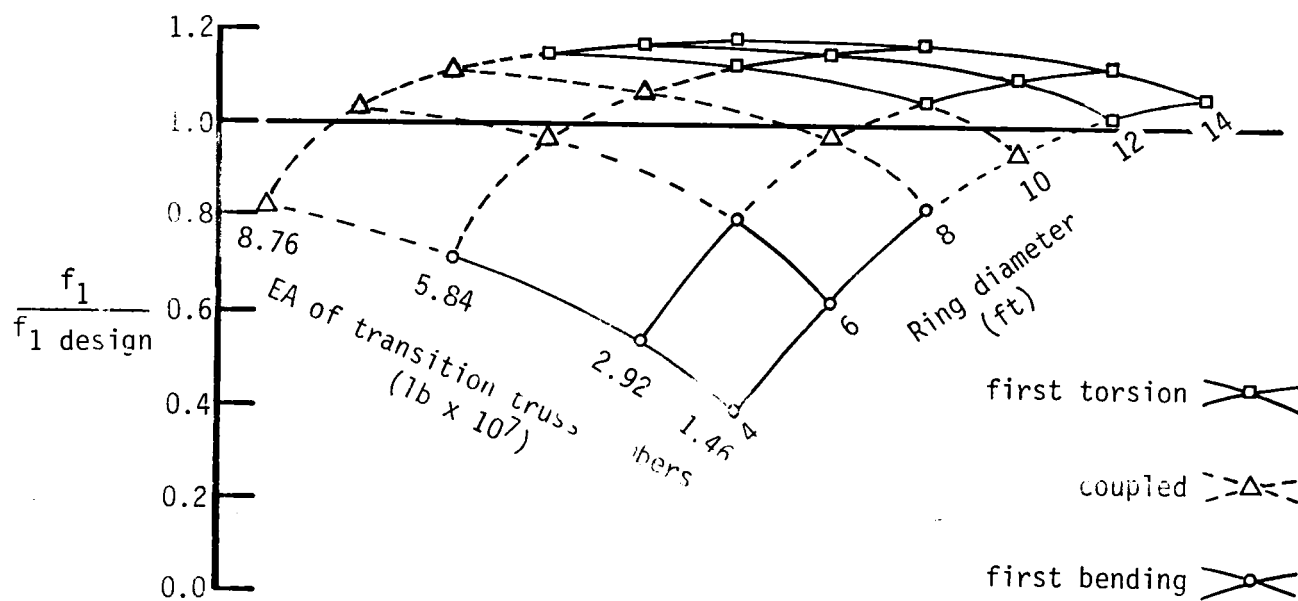


Figure 10a

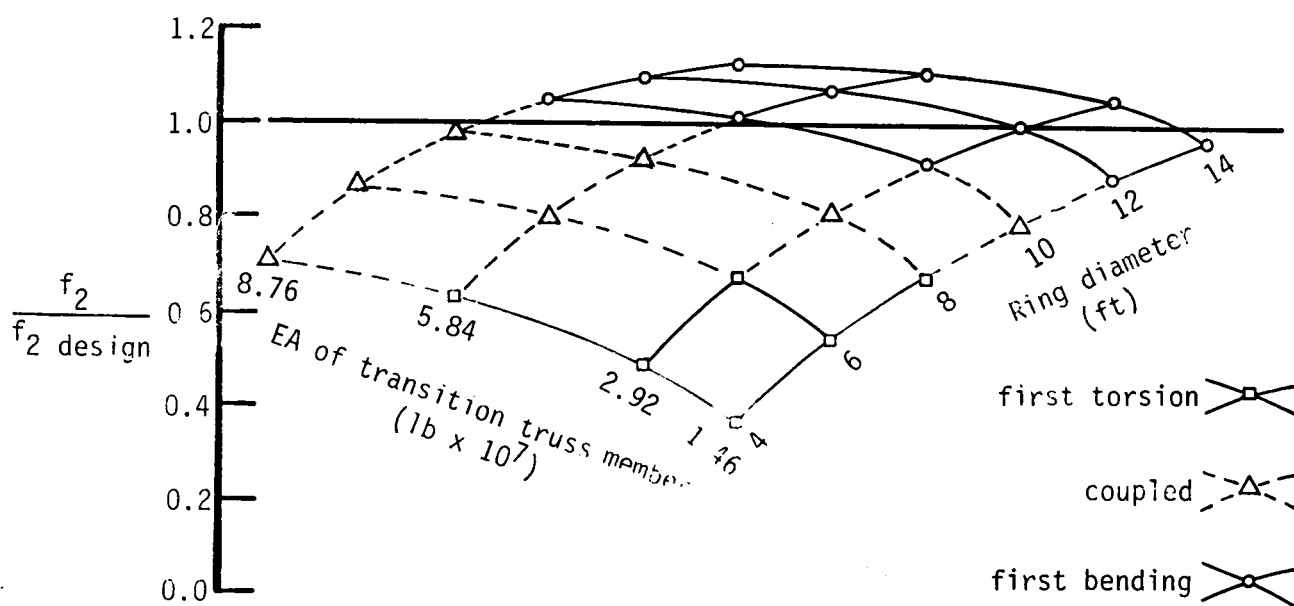


Figure 10b

Figure 10. Effect Of Transition Truss Member Stiffness And Ring Diameter

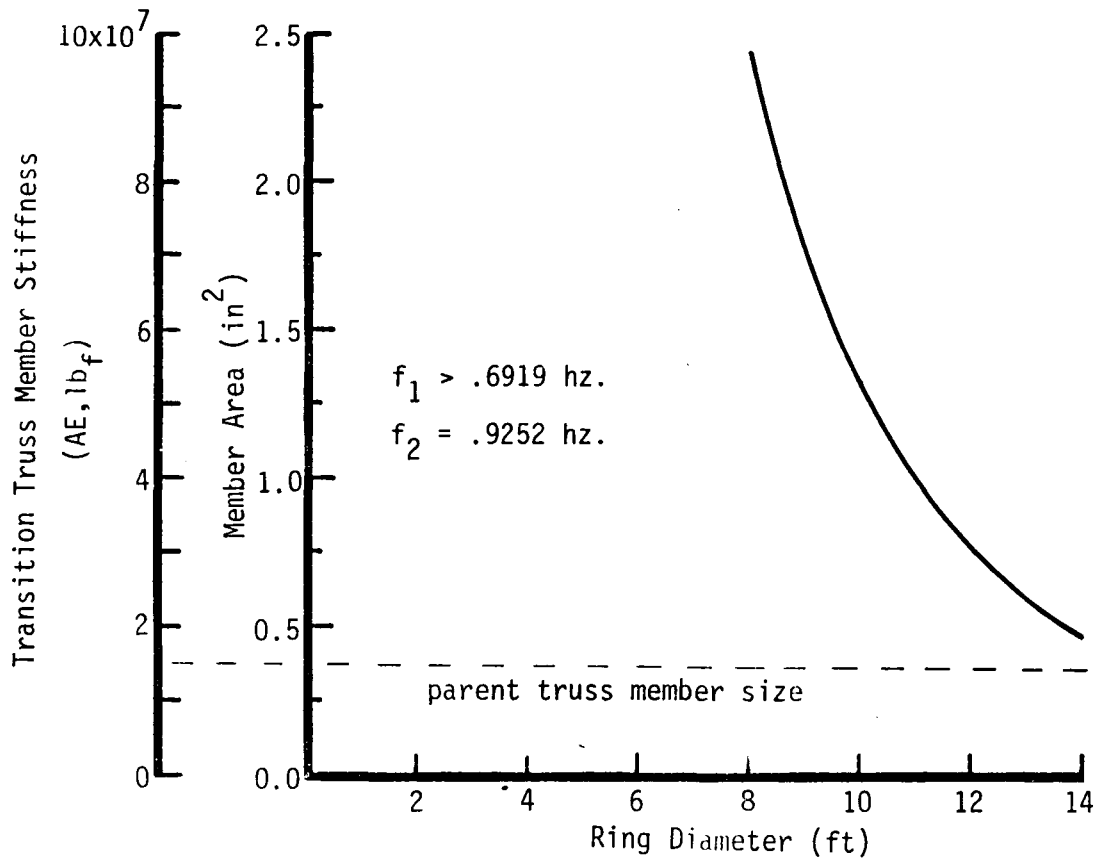


Figure 11. Design Curve For Transition Truss Member Size vs. Ring Diameter

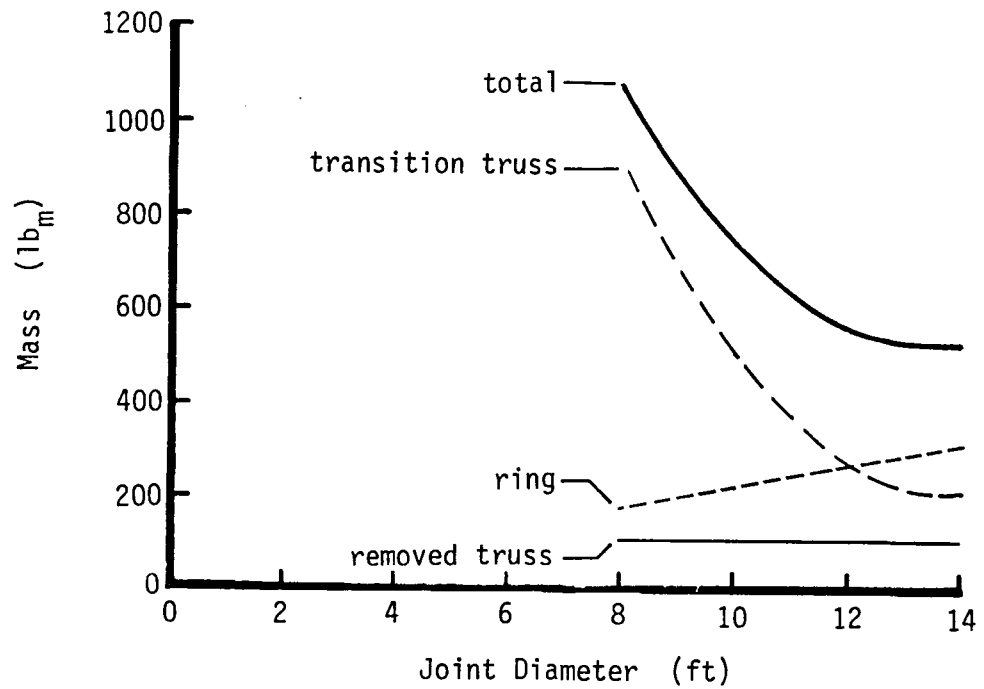


Figure 12. Rotary Joint Structural Mass

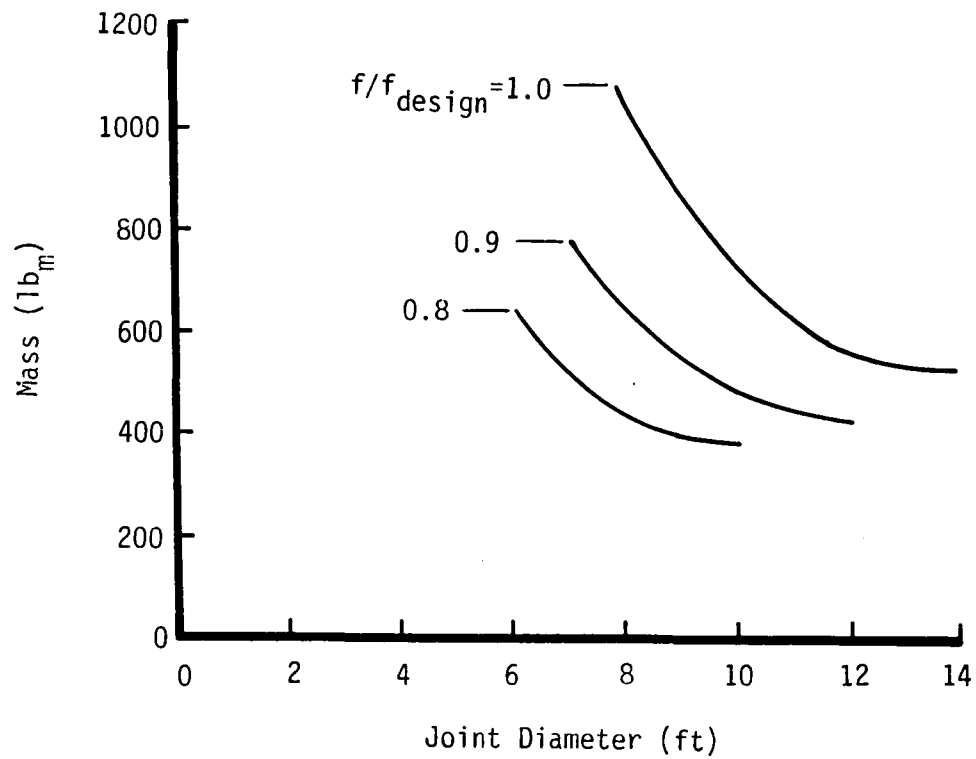


Figure 13. Rotary Joint Structural Mass for Reduced Joint Stiffness

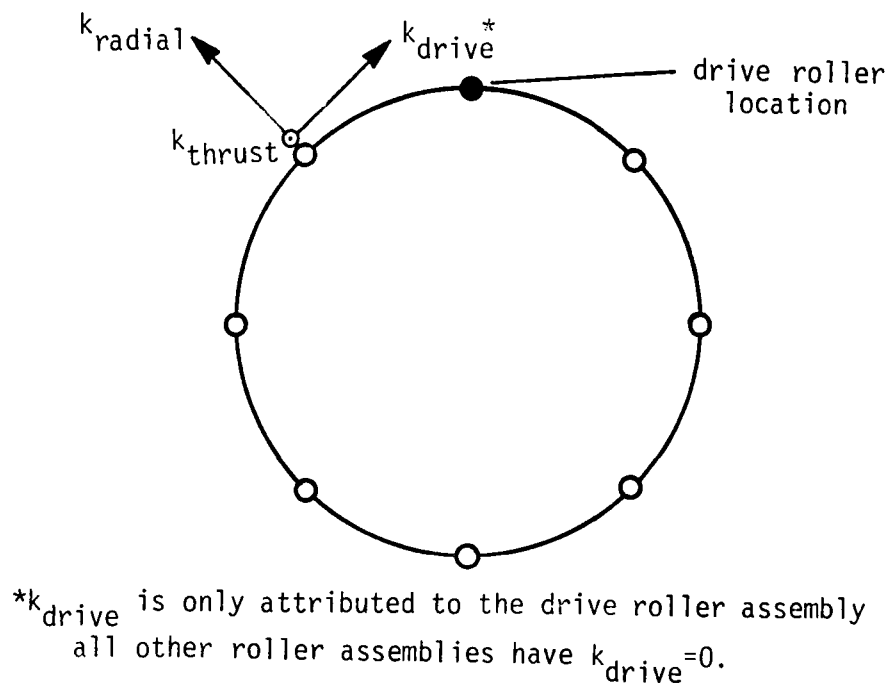


Figure 14. Roller Assembly Stiffness Components

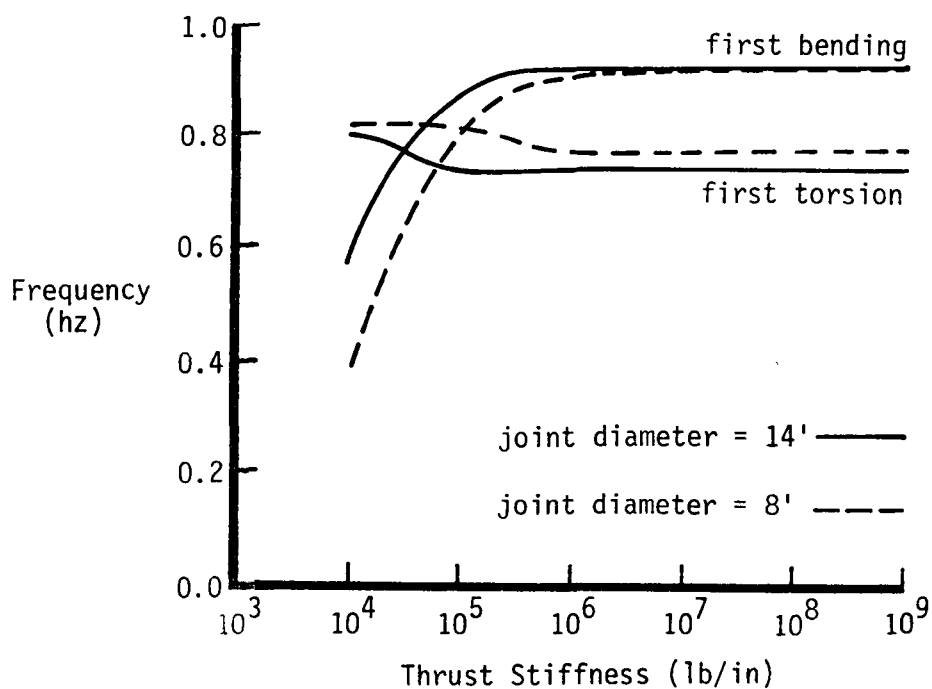


Figure 15. Roller Assembly Thrust Stiffness

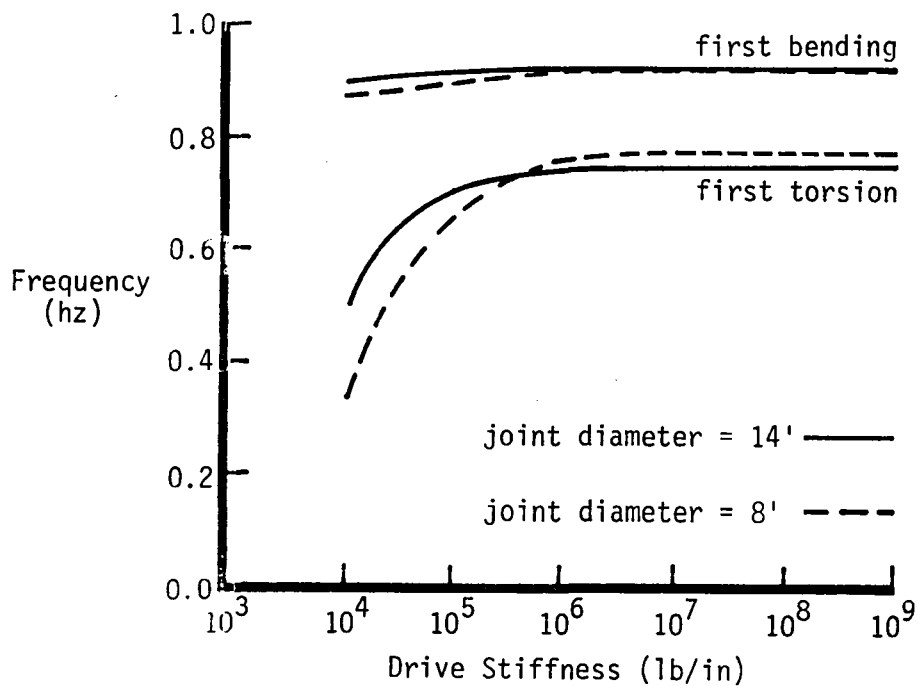


Figure 16. Roller Assembly Drive Stiffness

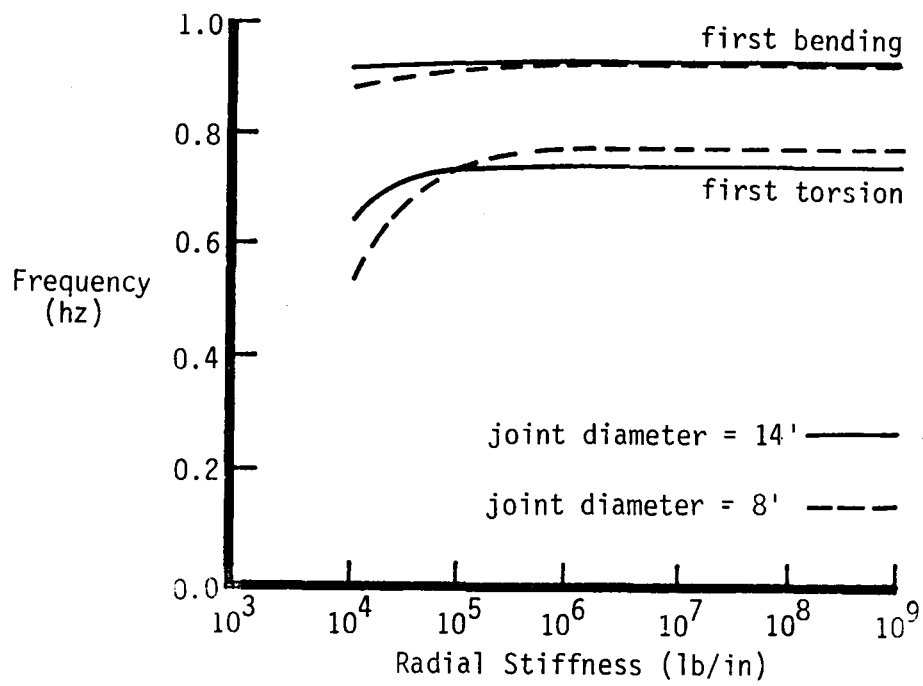


Figure 17. Roller Assembly Radial Stiffness

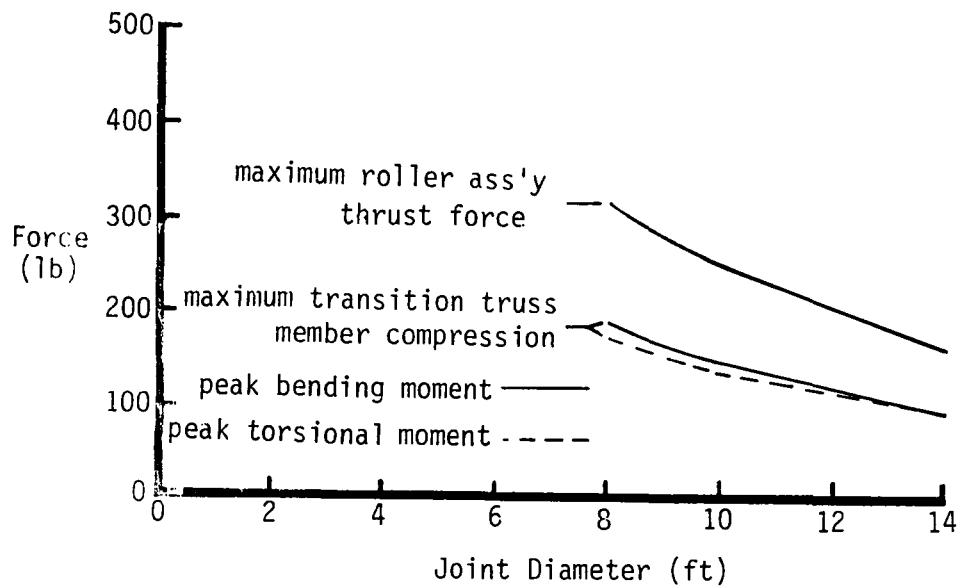


Figure 18. Maximum Internal Forces in Rotary Joint

APPENDIX A

Eccentricity in the Annular Ring Cross Section

In modeling the rotary joint it was assumed that the truss attachment points on the annular ring coincide with the elastic axis of the ring, or the ring is loaded non-eccentrically. While this may not necessarily be the case in a final design of the ring, this assumption can be used for designs in which the eccentricity is within a certain range.

Referring back to the internal strain energy method for curved beam analysis a similar calculation can be performed for the ring with eccentricity (h). Figure A-1 shows a diagram of the ring cross section with eccentricity indicated. A positive value of h represents a load path which lies radially outside of the ring's elastic axis. Computing the static deflection of this ring in general results in:

$$\Delta_p = PR^3 \left\{ \frac{2.6 \times 10^{-3}}{EI} + \frac{4.1 \times 10^{-5}}{GJ} \right\} +$$

$$P \left\{ \frac{(5.2 \times 10^{-3}R^2h + 2.6 \times 10^{-3}Rh^2)}{EI} + \frac{(5.2 \times 10^{-3}R^2h + 2.02Rh^2)}{GJ} \right\}$$

which reduces to equation (5) from the Analysis of the Annular Ring section for h equal to zero.

If the properties of the annular ring being considered are inserted into this equation (EI, GJ, R = 7.5 ft) and a load of one pound is assumed, then the equation reduces to a quadratic in h. This solution is plotted in figure A-2. In the region $-3" \leq h \leq 3"$ (i.e.- the depth of the ring) the quadratic function is fairly flat with the value of the deflection varying by only about 25 percent. Analysis of a proposed design should definitely include consideration of eccentricity effects. However, the results shown in Figure A-2 indicate that

eccentricity of ring loading (within the indicated range) would not significantly affect the study results (i.e.- Figures 9-18).

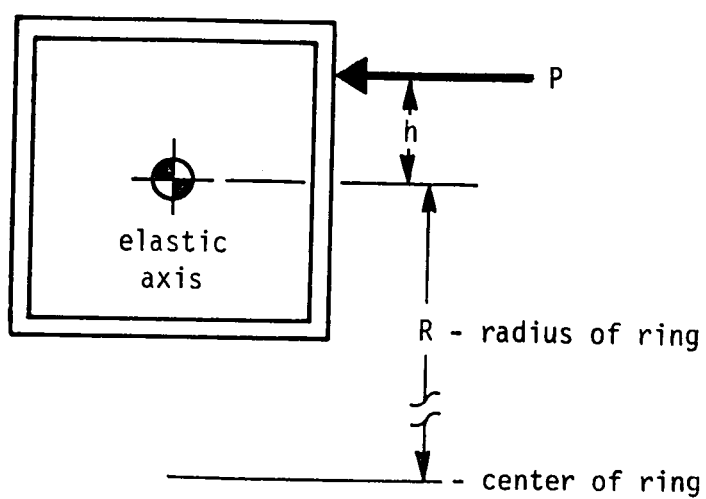


Figure A-1. Ring Cross Section With Eccentricity

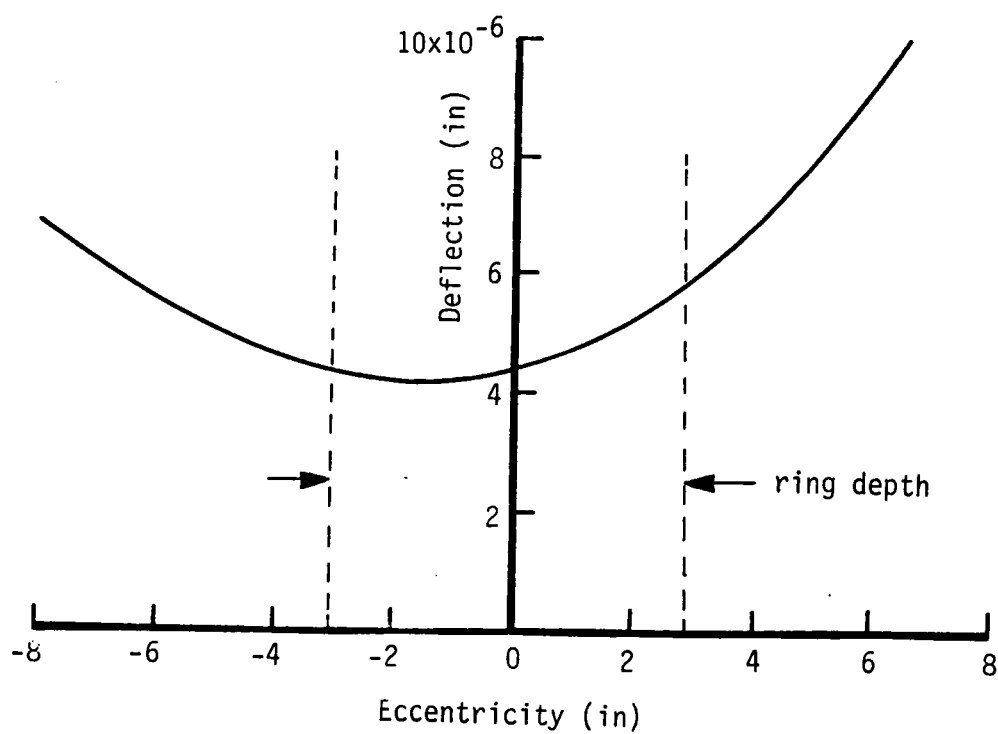


Figure A-2. Effect of Eccentricity on Ring Deflection

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16. Abstract <p>A study was conducted to define an annular ring, discrete roller assembly concept for the Space Station transverse boom rotary joint. The concept was analyzed using closed-form and finite element techniques, to size structural members for a range of joint diameters and to determine necessary equivalent stiffnesses for the roller assemblies. Also, a mass study of the system was conducted to determine its practicality, and maximum loads in the joint were identified.</p> <p>To obtain the optimum balance between high stiffness and low structural mass in the design of the rotary joint, it is necessary to maximize the diameter of the annular ring within operational constraints (i.e., shuttle cargo bay size). Further, a rotary joint designed with the largest possible ring diameter will result in minimum operational loads in both the roller assemblies and the transition truss members while also allowing minimum design stiffnesses for the roller assemblies.</p>					
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